Naval Surface Warfare Center
Carderock Division
West Bethesda, MD 20817-5700

Standardized Procedure for Experimental Vibration Testing of Damping Test Specimens

by

Colin P. Ratcliffe
United States Naval Academy, Mechanical Engineering Department

Roger M. Crane
NSWCCD, Structures and Composites Department

Dean Capone and Kevin Koudela
Applied Research Laboratory, Pennsylvania State University

19990520 004

DTIC QUALITY INSPECTED 4

DEPARTMENT OF THE NAVY

NAVAL SURFACE WARFARE CENTER, CARDEROCK DIVISION
9500 MACARTHUR BOULEVARD
WEST BETHESDA MD 20817-5700

9078
Ser 65-35
5 Apr 99

From: Commander, Naval Surface Warfare Center, Carderock Division
To: Chief of Naval Research (ONR 332)

Subj: VIBRATION TESTING OF DAMPING TEST SPECIMENS

Ref: (a)   Program Element 0602234N, Seaborne Structural Materials Program for FY96 and Program Element 0602121N, C-Section Task of the BA2 Surface Ship Hull, Mechanical, and Electrical Technology Program for FY96

Encl: (1)   NSWCCD-65-TR–1998/21, *Standardized Procedure for Experimental Vibration Testing of Damping Test Specimens*

1. Reference (a) requested the Naval Surface Warfare Center, Carderock Division (NSWCCD) to develop procedures to determine the modal damping of various material and geometric configurations.   Enclosure (1) presents a proposed standard for the experimental determination of structural and material energy dissipation, or damping.

2. Comments or questions may be referred to Dr. Roger M. Crane, Code 6553; telephone (301) 227-5126; e-mail, crane@dt.navy.mil.

J. E. BEACH
By direction

Subj: VIBRATION TESTING OF DAMPING TEST SPECIMENS

Copy to:

COMNAVSEASYSCOM WASHINGTON DC
  [SEA 03M3 (Ingle), SEA 03M, SEA 03P1 (Kadala),
  SEA 03P4, SEA 03R, PMS400D,
  PMS500 (Camponeschi)]

PEOSUB WASHINGTON DC [Spero]

NRL WASHINGTON DC [Code 6383 (Badaliance,
  Gauss)]

CNR ARLINGTON VA [ONR 332 (Kelly), ONR 334
  (Barsoum, Fishman, Rajapakse, Tucker,
  Vasudevan)]

USNA ANNAPOLIS MD [Mechanical Engineering
  Department (Dr. Colin P. Ratcliffe)]

DTIC FORT BELVOIR VA

Applied Research Laboratory
Pennsylvania State University
Attn: Drs. Kevin Koudela, Dean Capone
P.O. Box 30
State College, PA 16804

Assistant Director, CCM
Composites Manufacturing Science Laboratory
Attn   Dr. John W. Gillespie, Jr.
University of Delaware
Newark, DE 19716

Newport News Shipbuilding & Drydock Company
Attn   Dr. Tom Juska
4104 Washington Avenue
Newport News, VA 23607

NAVSURFWARCEN CARDEROCKDIV
  BETHESDA MD [Codes 0112 (Douglas),
  0115 (Messick), 3442 (TIC), 60 (w/o encl), 65,
  65R (2), 652, 652 (Young), 655, 6551 (Bartlett,
  Kiviat, Macander, Paraska), 6552 (Bonanni, Loup,
  Palmer, Rasmussen, Telegadas), 6553 (Crane (10),
  Coffin, Williams), 72 (Fisher), 7250 (Gershfeld,
  Maga, Warwick)]

2

Standardized Procedure for Experimental Vibration Testing of Damping Test Specimens

by

Colin P. Ratcliffe
United States Naval Academy, Mechanical Engineering Department

Roger M. Crane
NSWCCD, Structures and Composites Department

Dean Capone and Kevin Koudela
Applied Research Laboratory, Pennsylvania State University

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

Public reporting burden for this collection of information is estimated to average 1 hour per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding this burden estimate or any other aspect of this collection of information, including suggestions for reducing this burden, to Washington Headquarters services, Directorate for Information Operations and Reports, 1215 Jefferson Davis Highway, Suite 1204, Arlington, VA 22202-4302, and to the Office of Management and Budget, Paperwork Reduction Project (0704-0188), Washington, DC 20503.

1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE December 1998	3. REPORT TYPE AND DATES COVERED Final

4. TITLE AND SUBTITLE Standardized Procedure for Experimental Vibration Testing of Damping Test Specimens	5. FUNDING NUMBERS PE 0602234N PE 0602121N
6. AUTHOR(S) Colin P. Ratcliffe (USNA), Roger M. Crane (NSWCCD), Dean Capone and Kevin Koudela (ARL, Penn. State U.)	

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) Naval Surface Warfare Center Carderock Division 9500 MacArthur Boulevard West Bethesda, MD 20817-5700	8. PERFORMING ORGANIZATION REPORT NUMBER NSWCCD-65-TR–1998/21

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) Chief of Naval Research (ONR 332) Ballston Tower One 800 North Quincy Street Arlington, VA 22217-5660	10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT Approved for public release; distribution is unlimited.	12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

The experimental determination of structural or material damping as a function of frequency can be a complicated task. The observed values can apparently change between experiments. This report includes a proposed standardized test procedure which aims to reduce these variations in modal damping. The standard focuses on the four main areas where variations in test procedure can introduce differences: a) preparation and support of the structure; b) selection, preparation and use of transducers and equipment; c) setting up the analyzer, and d) data analysis and reduction.

14. SUBJECT TERMS structural damping material damping			15. NUMBER OF PAGES 27
			16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT UNCLASSIFIED	18. SECURITY CLASSIFICATION OF THIS PAGE UNCLASSIFIED	19. SECURITY CLASSIFICATION OF ABSTRACT UNCLASSIFIED	20. LIMITATION OF ABSTRACT SAR

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102

Contents

Figures

Tables

Administrative Information

1. Introduction

1.1 This report presents a proposed standard for the experimental determination of structural and material energy dissipation, or damping. When damping is required as a function of frequency, the levels of damping have traditionally been determined by dynamic testing. These tests are often a modal analysis of a structure or damping test component, such as a flat plate. While modal analysis is a well-established test procedure, there is no published standard that ensures consistency between different tests or different establishments. This proposed standard defines the experimental modal procedures for finding the amount of damping in flat plate damping test pieces.

1.2 The standardized procedure presented here is based on studies conducted in collaboration between the United States Naval Academy, the Naval Surface Warfare Center, Carderock, and the Pennsylvania State University Applied Research Laboratory.

2. Background and Scope

2.1 This standard covers the vibration testing of small plates used to determine the modal damping of various material and geometric configurations. Small in this context typically means the thickness of the plates is in the 1/8- to 3-inch range, and the length and width are both less than about 36 inches. However, the procedures and discussion in this standard are also applicable to the modal analysis of larger and geometrically more complex structures. The primary focus of this standard is to ensure that when vibration testing, the data obtained during the experiment are suitable for making accurate modal damping estimates.

2.2 The data obtained from experimental vibration testing of flat plates can yield apparently different levels of damping, depending upon the test configuration and test procedures. In order to use modal analysis to obtain good estimates of damping, the primary requirement is for high quality frequency response function data. To obtain these data, several procedural steps and decisions have to be taken. These are: (1) Choice of test technique; (2) preparation and support of the plate; (3) calibration of transducers; (4) analyzer setup and; (5) data capture. Details of these steps are provided in the following paragraphs. The standard does not cover the actual numerical analysis of the data.

3. Experimental Procedure

3.1 Choice of Test Method

3.1.1 This section discusses the overall aspects of the experimental procedure. Procedural details are covered later. As with all experimental tests, there can be a significant amount of iteration in the setup before the final data are collected. This section does not investigate such iteration.

3.1.2 When damping estimates are of primary importance, it is critical that the instrumentation and test method have minimum interference with the structure under test. Stingers (connecting rods) typically used to connect electrodynamic exciters to structures do not provide the necessary degree of isolation. They adversely influence the structure by changing the local stiffness and energy dissipation. This means that test techniques using electrodynamic exciters usually will not provide the highest accuracy damping estimates. Currently, the only vibration test method that offers the necessary isolation is impact testing. In this method, the dynamic frequency response functions between pairs of test grid points are obtained by impulsively exciting the structure with an instrumented hammer, and measuring the response with an accelerometer at a fixed reference point. The hammer includes a force gage to measure the excitation. Using this method, the only interaction between the equipment and the structure is the impact of the hammer on the structure (desired and measured), and any undesirable effects of the accelerometer. The primary undesirable accelerometer effect is mass loading. This effect can be minimized by choosing a lightweight accelerometer. A discussion on the selection of transducers and other equipment is in Section 5.

3.1.3 The preferred test method is therefore impact excitation, with a single fixed point reference response accelerometer. Testing with multiple reference accelerometers is acceptable to this standard, but is not covered here.

3.2 Overall Experimental Procedure

3.2.1 Prior to the capture of frequency response functions, transducers will be selected and calibrated. The test plate will be marked with the test grid, the plate will be supported, and have a reference accelerometer secured to it. The analyzer will then be set up.

3.2.2 When PCB ICP signal conditioning amplifiers are used, they will be switched on at the start of the test. To ensure filters have settled to their steady-state conditions, they must be left on for at least one minute before final data are captured. Before testing, it should be verified that there is sufficient battery power to last the entire test. Changing batteries during a test should be avoided if at all possible. For analyzers with ICP inputs, the manual should be consulted to verify that the transducer dc offset has settled before commencing data capture. The setting time will usually be at least 30 seconds, and can be up to more than one minute.

3.2.3 The frequency response function will be measured between each test grid point and the reference accelerometer. For each grid point, there will be several repeated measurements, averaged to determine the final data. For "well behaved" structures, there will be at least three

separate measurements for the averaging. When the data capture is more problematic, the number of averages will be increased. Providing the experiment is set up correctly, and the structure is amenable to modal analysis, it should not be necessary to increase the number of averages above six. When six averages are deemed insufficient, this fact should be documented in the project report.

3.2.4 During the measurement, the analyzer will be set to show, as a minimum, the frequency response function (log magnitude) and the coherence. There can be a number of times when data measurement for a grid point has to be repeated. The following are situations when repeating a measurement is necessary:

a)   When anyone involved in the test detects an abnormal hit, even if there is no apparent degradation of the frequency response function or coherence. Mis-hits are common, and can be detected by the "feel" of the hammer, or by listening to the hit. Typical causes of mis-hits are "double" hits, and the hammer head not being aligned at 90° to the surface. This latter problem can cause sliding of the hammer tip on the surface of the structure, and shear and bending in the force gage. All of these effects can degrade the data.

b)   When the frequency response function shows any kind of abnormal rippling or picket fence effect. Providing the equipment is correctly set up, ripples on the frequency response function normally indicate a poor hammer hit.

c)   When the coherence indicates poor correlation between the data sets measured during the averaging process. The coherence function for impulse testing gives a very good feedback for the overall quality of the test. Coherence will be degraded by many problems, including structural nonlinearity, poor signal-to-noise ratio, poor hammer technique, hitting at slightly different locations, and transducer problems such as the accelerometer coming loose. A good coherence does not guarantee excellent data. However, poor coherence is nearly always indicative of poor data. Figures 1 to 4 give examples of typical coherence functions.
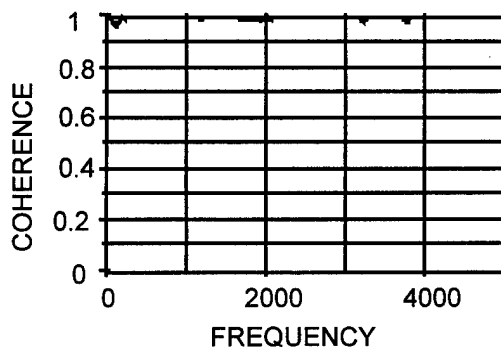
Figure 1. An example of good coherence.
The data for the frequency response
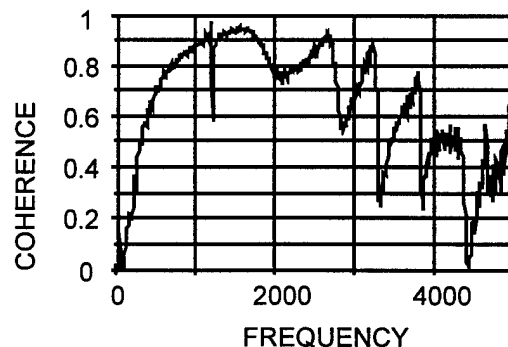function can be used at all frequencies.

Figure 2. This coherence function
indicates problems with measurement.
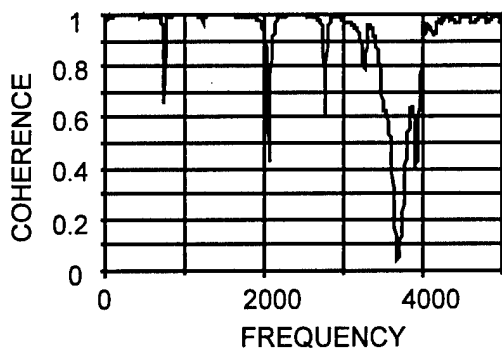The measurement should be repeated.

Figure 3. Sharp dropouts near antiresonances are normal, and do not affect the modal damping estimates. The large dropout near 3800 Hz is coincident with a dropout in the auto power spectrum of the force signal (see 5.1.1). This coherence indicates the frequency response function is probably acceptable, except near 3800 Hz.
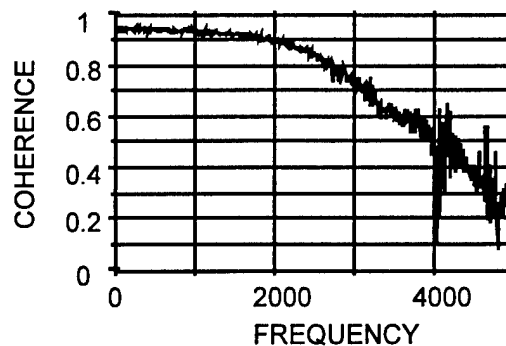
Figure 4. This coherence function is not typical of that obtained for impulse testing methods. The curve indicates that there might be large amounts of signal noise. The roll-off above about 2 kHz suggests there may be a filtering problem somewhere in the equipment, or other frequency related problem.

3.2.5 Once the frequency response functions for all the test grid points have been measured and transferred to computer, the experimental phase is complete.

4. Preparing and Supporting the Structure

4.1 The Test Grid

4.1.1 Theoretically the damping of the test structure could be determined from a single frequency response function measurement. However, estimates obtained this way have little statistical justification. Also, there would be no mode shape information, which it is sometimes useful to have when analyzing damping behavior. Therefore, this standard requires that damping estimates will be obtained from a modal analysis based on a number of frequency response functions, measured from several points across the structure. The final damping estimated by the analysis will thus be based on a large number of independent vibration tests. The modal analysis will also supply mode shape information.

4.1.2 The plates will be marked with a grid of numbered test points, as shown in Figure 5. There is some degree of flexibility in choosing suitable locations for the grid points, but it can be advantageous to use a pattern that is repeatable from test to test, and that obtains data representative of the whole structure. Therefore, the entire plate will be marked with a rectangular grid. The aspect ratio of the grid shape will not exceed 2:1. The uniform mesh will be 'shifted' such that there is a half-mesh gap between the edge grid points and the edge of the structure. This is to minimize edge effects when hitting the structure with the hammer. Hitting too close to the edge can cause anomalies in the data that are manifested as reduced quality mode

shapes and damping estimates. The grid size will be chosen to generate between 40 and 150 test points. Choosing a number in this range offers the best compromise between accuracy of damping and mode shape estimates, and measurement time. If it is essential to use less than 40 test points, the fact and reason for the choice must be documented in the project report. Using more than 150 test points is acceptable. However, using a very large number of test points does not guarantee an increase in the accuracy of the damping estimates, and the increase in data acquisition and analysis time is not normally justified. While the numbering sequence for the test grid is not critical, Figure 5 shows a typical layout that is compatible with many commercial modal analysis packages.

4.1.3 Commensurate with the previous discussion on the mesh shape and size, each dimension of the mesh should normally be at least 1/2 inch, and less than 3 inches. The grid details should form part of the project report.
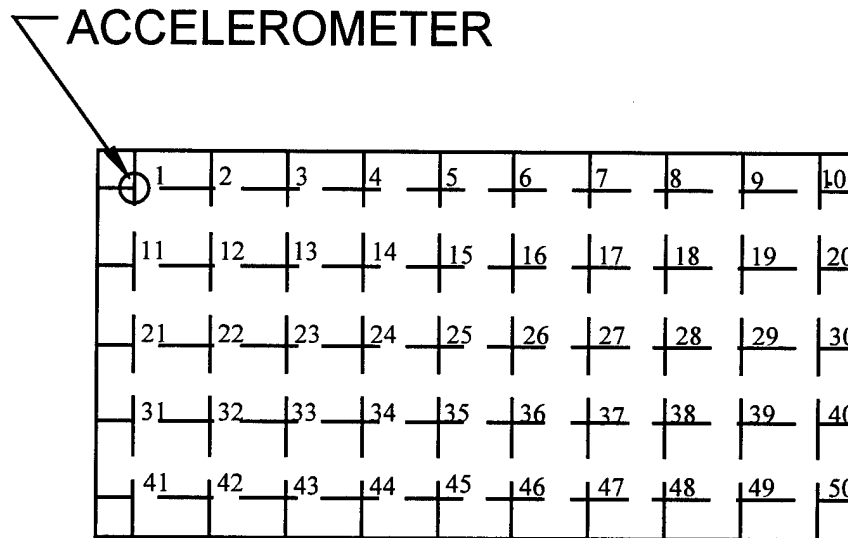
Figure 5. Test Grid.

4.2 Structure Support - General

4.2.1 The plate should be supported such that there is minimum interaction between the plate and the supporting fixture. The fixture can interact with the plate in at least three ways. First, energy may dissipate into the fixture, and this will generally lead to increased damping estimates. Energy dissipation like this is difficult to replicate between tests. As a result, ranking of structures based on their levels of damping can be very difficult. Second, any resonances of the fixture itself will lead to erroneous data. Third, the fixture will effect the boundary conditions of the structure. This may cause the natural frequencies, mode shapes and energy dissipation to change.

4.2.2 There are two different philosophies for designing a suitable support fixture. The first strategy is to build a very stiff fixture, whose resonances are well above those of the structure under test. This is the standard recommended approach used, for example, in some shock testing. A stiff fixture is not normally recommended when damping is the primary

quantity of interest. This is because the test structure has to be secured to the fixture, and the interface between the structure and fixture will dissipate energy through friction. The second fixture design, and the preferred method for this standard, is to provide very flexible supports. In this case, the main natural frequencies of the structure on the support should be much lower than the first structural resonance of the test structure. As a guide, the first structural resonance should be at least three times higher than the rigid body support resonances, and preferably ten times higher.

4.2.3 The plates will normally be supported horizontally, i.e. the length and width axes of the plate will be horizontal. When plates are suspended in any other orientation, the change in boundary conditions and reduced ability to obtain high quality data can result in degraded damping estimates.

4.2.4 For the damping test plates covered by this standard, there are two acceptable support methods. The two methods are rubber stoppers and bubble wrap. These support methods are discussed in Section 4.2 et. seq. Note that when a comparative study is conducted, it is preferable to use the same support method for all tests in the series.

4.3 Structure Support - Rubber Stoppers

4.3.1 Of the two acceptable methods, support on rubber stoppers is preferred. The stoppers, as shown in Figure 6, are of natural rubber with a Shore A hardness between 40 and 50. The base diameter, d, of the stoppers will be as small as possible, commensurate with being able to support the weight of the plate without undue deformation of the stoppers. Also, the diameter, d, and height, h, will be the same. Table 1 gives suggested stopper sizes, depending on the weight of the test plate. Unusual plate configurations may require a different number of stoppers.

Table 1. Choice of Supporting Rubber Stoppers.

Weight of test plate (lbs)	Number of rubber stoppers	Diameter, d and height, h (inch)
less than 1	4	1/8
1-10	4	1/4
10-20	4	3/8
20-50	6	3/8
50-100	6	1/2
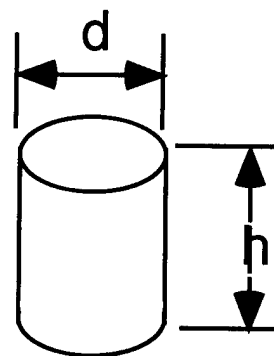
Figure 6. Rubber Stopper.

4.3.2 Support of the plate is achieved by placing the stoppers in the correct configuration on a hard, smooth surface such as concrete or steel. The plate is then placed on the stoppers. Optionally, the stoppers may be glued onto the plate and/or hard surface. Gluing provides for a more repeatable support pattern, and prevents the stoppers from moving out of place. Gluing to *both* the plate and hard surface is preferred, especially for lighter panels. The location of the

stoppers is as shown in Figure 7, where the width of the plate, W, is smaller than the length of the plate, L. The support pattern is chosen to minimize the effect on the major plate modes. When thin plates with a large aspect ratio are tested, there may be an unacceptable sag between the supports. If so, it may be necessary to use extra sets of stoppers along the length. Alternatively, the second support method (bubble wrap, Section 4.4) may be used.
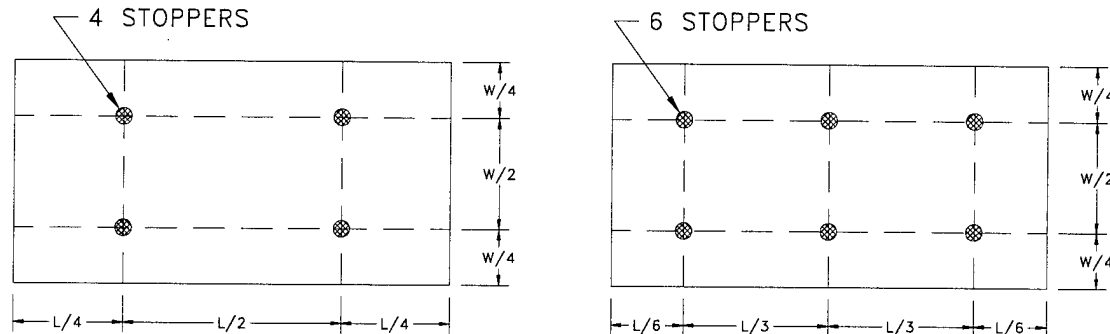
Figure 7. Stopper Pattern for 4 and 6 Support Points.

4.4 Structure Support - Bubble Wrap

4.4.1 The second acceptable support method is to lay the plate on commercial bubble wrap. The bubble wrap will be placed on a horizontal, hard, smooth surface such as concrete or steel. The bubble wrap will be placed with the "bubbles" upwards (continuous membrane downwards, next to the hard surface). A new piece of bubble wrap is to be used for each test. However, for repetitive tests of the same plate, it is preferable to leave the plate and bubble wrap untouched between tests.

5. Transducers

5.1 Preferred Transducers

5.1.1 The preferred method of excitation is with a modally-tuned force hammer, PCB ICP model 086B03 or equivalent. The choice of hammer tip depends on the resilience of the plate surface, and the frequency range of interest. The normal choice of tip is nylon or steel. A nylon tipped hammer is more forgiving of the tester, and is preferred when acceptable. However, there are some situations where a steel tip is preferred. The decision should be based on observing the auto power spectrum of the hammer signal. Energy should be present at all test frequencies. The short duration of the force signal causes a 'picket fence' effect in its auto power spectrum. In practice, it can be acceptable for the first dropout in this spectrum to be within the test frequency range. The energy dropout will cause a degradation of data quality near the dropout frequency (see, for example, Figure 3), and this dropout should be documented in the project report. Increasing the hardness of the hammer tip increases the frequency at which the dropout occurs. The modally tuned hammer can be used with or without the optional added masses.

5.1.2 The preferred response transducer is a PCB ICP accelerometer with a mass less than 2 grams. Individual modern accelerometers can normally be used over a very wide frequency range (typically less than 1 Hz up to many kHz), but it should be verified that the chosen accelerometer is rated for the frequency range of the test. The accelerometer will be secured to the test structure using beeswax or cyanoacrylate glue. A small amount of wax or glue is spread over the contacting surface of the accelerometer. The accelerometer is then pushed onto the surface of the structure, and given a small "twist." This process extrudes most of the wax or glue, leaving a very thin film that bonds the accelerometer tightly to the surface.

5.1.3 When comparative tests are conducted, the same physical transducer pair should be used for all tests if possible.

5.1.4 The signal conditioning for PCB ICP transducers will be with PCB ICP amplifiers. Battery powered amplifiers generally have less electronic noise, and are therefore preferred. For analyzers with an ICP input option, this option should be selected in lieu of using external ICP amplifiers. Manufacturers specifications should be followed with regard to signal levels, and the delay before starting measurements.

5.2 Calibration - General

5.2.1 There are three ways of calibrating transducers. In order of preference (most preferred first) these are: No calibration; Relative calibration; Absolute calibration. Each calibration method is discussed below.

5.2.2 **No Calibration.** Transducers do not have to be calibrated when a single pair of transducers is used for the entire test, and when the required results only include natural frequencies, relative mode shapes, viscous damping ratios, and loss factors. Although many different algorithms are used by different modal analysis packages, energy dissipation is essentially based on the half-power bandwidth. While the absolute level of the dynamic functions depends on calibration, the *relative shape* of the functions does not. Hence the half-power bandwidth (and resulting damping measurements) does not depend on having absolute values for the dynamic functions.

5.2.3 Also, any form of calibration is time consuming, and potentially expensive. Therefore, when acceptable to the project, using uncalibrated transducers is preferred to using any form of calibration. Test reports should indicate that the dynamic functions are referenced to an arbitrary datum.

5.2.4 **Relative Calibration.** Relative calibration of transducers is necessary and adequate when absolute values for the dynamic functions or the modal constants are required. Relative calibration means that, for example, the force signal divided by the accelerometer signal yields the correct value for a rigid calibration mass, but neither the force gage nor accelerometer provide absolute values for their measured quantities. The procedure for relative calibration is described in Section 5.3. When calibration is necessary, relative calibration is the preferred calibration method. Relative calibration usually provides for more accurate measurements of the frequency response functions than is possible with absolute calibration.

5.2.5 **Absolute Calibration.** Absolute calibration of transducers is rarely required, and should only be considered when essential to the project. Absolute calibration is only necessary when the absolute level of excitation or response is required. This may be, for example, when

the structure is nonlinear, and the test level is to be recorded. Impact testing and modal analysis assume linearity and are not normally used for nonlinear testing. Absolute calibration, and tests requiring absolute calibration, are beyond the scope of this standard.

5.3 Relative Calibration Procedure

5.3.1 This section will be included in Revision 1 of this standard.

6. Analyzer Setup

6.1 Background

6.1.1 This standard does not present the complete setup necessary for any particular spectrum analyzer, and it is assumed that the tester is familiar with the details of their instrumentation. Different products use different terminology for various settings. This standard discusses the critical analyzer settings in a generic form.

6.2 Signal Input

6.2.1 The conditioned time history signals for the force excitation and acceleration response are captured and digitized with a digital spectrum analyzer. The required output from the analyzer is a frequency response function. Therefore, the analyzer has to be initialized to process two channels of data, and to provide the necessary frequency domain complex division. Typically, two-channel analyzers divide the data for Channel 2 or Channel B with that measured for Channel 1 or Channel A. For analyzers complying with this arrangement, the conditioned force signal is to be connected to Channel A, and the conditioned accelerometer response signal is to be connected to Channel B.

6.3 Frequency Range

6.3.1 The frequency range may be given in the project specifications. When the frequency range is not specified, the following should be taken into consideration:

a) Typically, the damping tests covered by this standard aim to provide the level of modal damping as a function of frequency. Therefore, it is necessary to measure several natural frequencies in order to obtain the necessary information.

b) As a rule of thumb, it is desirable to measure at least the first five lengthwise bending modes of the test specimens. However, without conducting a modal survey, it can be difficult to differentiate between the different types of mode. Therefore, in a "blind" test, the frequency range should be adjusted to encompass the first 10-20 modes. This will normally ensure sufficient bandwidth to include the required bending modes, without unduly degrading the frequency resolution.

c) An indication of the number of resonances in any given bandwidth can usually be obtained by taking "rough" measurements using the analyzer's default settings, and looking for peaks in the frequency response functions. While these data will be of

9

poor quality, are unreliable, and should not be saved to computer, the frequency response functions can be observed on the analyzer screen. Interactively changing the frequency range, and taking these rough measurements will generally provide sufficient information to enable a good choice of frequency range.

d)   Unless dictated by the project specifications, the frequency range should usually start at zero (i.e. baseband data). Zoomed data may be captured only when essential, for example, because of insufficient frequency resolution.

e)   Care should be taken to ensure the frequency range is not set too high. Above about the 20th resonance (this figure varies significantly for different types of structures), it is increasingly difficult to define the dynamic behavior of a structure as 'modal.' Hence, high frequency data collected as part of the modal damping test will not be used for the modal analysis. This unnecessary high frequency data represents 'lost' frequency resolution for the lower ordered resonances.

6.4 Input Coupling

6.4.1 The signal conditioning amplifiers for the preferred transducers have a dc offset. Therefore, all inputs to the analyzer must be ac coupled to remove this offset. If non-preferred transducers are used, the input should also be ac coupled, unless there is a specific reason for not doing so. For analyzers with an ICP input option, this option should be used. The analyzer instruction manual should be consulted to ensure optimum settings.

6.5 Input Gain

6.5.1 When transducers are not calibrated, the input gain should be set to volts/(engineering unit). For the preferred transducers, the value to enter for each channel is the gain (x1, x10, x100) set on the PCB ICP conditioning amplifier.

6.5.2 When transducers have been calibrated (relative), the input gain for each channel should be set to the values determined from the calibration.

6.6 Input Range

6.6.1 The input range defines the maximum voltage that can be accommodated by the analyzer's analog to digital converters. In order to obtain data with a minimum of digitization noise, the input ranges have to be set as small as possible, while being large enough to accept the maximum signal voltages without overload. For most digital spectrum analyzers, the 'auto' input range option is inadequate for impact testing. Manually setting and maintaining the levels usually provides for better control of data quality.

6.6.2 The input range for each channel must be set independently. Both levels are set in a similar fashion. The analyzer is set such that it indicates an input signal overload. On some analyzers, this indication is always functional (e.g. a flashing LED indicates overload, even when the instrument is not actually recording a measurement.) On other analyzers, a measurement has to be 'started' before the overload indication is functional. The input range is initially set very low. The plate is then repeatedly impacted at various locations across its surface. The analyzer input range is increased until the analyzer no longer shows any overload indication. The input

range is then increased by approximately 10-20% (hardware dependent). This increase allows for some variation in the strength of the impact, and other structurally dependent variations, while not unduly degrading the data with digitization noise.

6.6.3 Alternatively, some analyzers have 'overload' and '50%' lights. The input range should be set such that for each channel the overload light does not illuminate, while the 50% light does. This ensure optimum use of the analog to digital converters.

6.6.4 For the preferred transducers, and when using PCB ICP signal conditioning amplifiers, the input range determined using either of the above procedures should ideally be between 0.5 and 7 Volts. If the range is outside these limits, the gain on the conditioning amplifiers should be increased or decreased as appropriate, the input gain changed if necessary, and the input range determined again. Under no circumstances should the input range be set higher than 9 Volts, since this will indicate the transducers and conditioning amplifiers are operating above their optimum linear range. The analyzer input range may be reduced below the 0.5 Volt limit if signal levels are small. However, the project should be documented that there may be a higher than normal amount of signal noise in the data. For analyzers with an ICP input, there is not usually a separate input gain control. This means signals may be lower than 0.5 Volts without being low grade. The analyzer manual should be consulted to verify signals levels are adequate.

6.6.5 For all other analyzers, transducers and signal conditioning equipment, the tester must determine that the analyzer input range is within normal tolerances.

6.6.6 During the test, there will normally be some variation in signal strength from impact to impact, and test coordinate to test coordinate. It is preferred that, if possible, input ranges and conditioning amplifier gains are not changed during a test. It is generally preferable to vary slightly the impact force, and maintain the input range settings, rather than change amplifier and input gains. The small variations in data caused by hitting the structure harder or softer, are usually less than the variations caused by changing the input range and/or amplifier gains.

6.6.7 When available on the analyzer, overload reject should be enabled. This feature automatically discards any data that causes an overload of the analog to digital converters. Some analyzers have this feature permanently enabled.

6.7 Pretrigger

6.7.1 Data capture will be triggered from the force hammer signal. For frequency ranges up to 2.5 kHz, there will be a pre-trigger of 10 ms. For frequency ranges above 2.5 kHz, a pre-trigger of 5 ms is generally adequate. The pre-trigger should never be set less than 5 ms. It should be verified, by observing sample impacts in the time domain, that the selected pre-trigger is sufficient. The signal should show a small amount of time trace before the impact signal starts. Both the force and accelerometer channels must have the same pretrigger.

6.8 Windowing, Etc

6.8.1 Windowing of the time signals enhances the performance of the Fourier transform algorithm used in the analyzer by reducing leakage. Separate windows are necessary for the force and response signals. For consistency, when a series of tests is conducted, the window functions should be the same for each test if acceptable or possible.

6.8.2 The actual force signal is zero (plus signal noise) after the end of the impact. This is because the hammer is no longer in contact with the surface of the plate. Therefore a square window whose characteristics are [1 ... 1, 0 ... 0] will be used for the force signal. The width of the window (i.e. the time for which the window has a value of 1) will be set such that the step down to zero occurs between 10 and 20 ms after the end of the impact. Selecting this length of window ensures all of the force signal is captured correctly, while offering maximum windowing effects. Some analyzers have a square window whose characteristics are [0 ... 0, 1 ... 1, 0 ... 0]. When this window is used, care should be taken to ensure the step up from the initial zeros to the ones occurs well before the start of the hammer's impact. For analyzers that do not have rectangular windows of the type described in this paragraph, an exponential window (described below) should be used. In this case, a very sharp window should be used.

6.8.3 The response signal has the overall characteristics of a negative exponential function, and most of the important information is in the first part of the trace. Therefore, an exponential window is preferred for the response signal. The aim of using this window is to 'taper off' the response signal such that it is very small at the end of the captured time. Exponential window functions can defined by either a time constant or an attenuation. When the window is defined by the equation $e^{-t/\tau}$ then τ (in seconds) is the window time constant set in the analyzer. When a window is defined by an attenuation, the attenuation, a, is the ratio of the window amplitude at the end of the window, divided by the window amplitude at the start of the window. For windows with the same effect, the time constant and attenuation are related by:

$$e^{-T/\tau} = a$$

where T is the length (seconds) during which the window is applied. Some analyzers start the window at the start of the time record, and finish applying the window at the end of the time record. Other analyzers offer many variations on how and when the window is applied and defined.

6.8.4 The amount of windowing required depends on whether the structure is heavily or lightly damped. Recall that the aim of the window is to 'taper off' the response signal by the end of the captured time. For heavily damped structures, the natural decay of the vibration itself may be sufficiently large that this requirement is automatically met. In this case, the exponential window can have a long time constant. For more resonant structures, a sharper window is required. Conversely, if too short a time constant is used, the response signal will unnecessarily be reduced, and potentially there will be a degraded frequency response function. When considering the transient response of a structure, the terms "heavy damping" and "light damping" depend on the product of the viscous damping ratio, the natural frequency, and the length of the time record. For structures whose damping is relatively independent of frequency, the lowest natural frequency is often the mode that controls the window requirement, in which case the following can be used as a guide:

Heavy damping implies that $(\zeta_r f_r T) > 1$

Light damping implies that $(\zeta_r f_r T) < 0.1$

where ζ_r is the viscous damping ratio, f_r is the natural frequency in Hz, and T is the length of the window in seconds. We can normally assume that, when a signal has attenuated by -75 dB, it

has essentially reduced to zero. For a lightly damped structure (as defined above), the natural attenuation of the signal during time T is approximately -5 dB. The exponential window therefore needs to attenuate the signal an additional -70 dB. For a heavily damped structure, the natural attenuation of the signal during time T is approximately -55 dB. The exponential window therefore needs to attenuate the signal an additional -20 dB.

6.8.5 The nomograph in Figure 8 is used to determine the required exponential window time constant. The "Damping Factor" on the ordinate of Figure 8 is the product ($\zeta_r f_r T$). Therefore, some knowledge of the structure and frequency range is required before the set up of the analyzer can be completed correctly. Reasonable estimates of ζ_r and f_r are generally adequate. The "Window Length" shown as the abscissa in Figure 8 is the time during which the response signal is windowed. Typically this time length would be the same as the length of the captured time record, and depends on the frequency range and number of data points in the Fourier transforms.

6.8.6 The setting for some analyzers is an attenuation, rather than a time constant. The nomograph in Figure 9 can be used to convert between an exponential time constant and an attenuation.

6.8.7 The accelerometer window time constant (not the attenuation) is needed for a correction that must be applied to the modal data. Therefore, the window time constant must be recorded in the project report. An unusual test or structure may require an exponential window setting outside the range given by the Figures 8 and 9. In this case, the reason for the choice should be documented.

6.9 Averaging

6.9.1 Each archived frequency response function will be the result of averaging several frequency response functions obtained from repeated measurements at each test grid point. The number of such measurements was discussed in Section 3.2.3. The frequency response functions will be averaged in the frequency domain. Averaging will be stable mean, i.e. linear averaging.
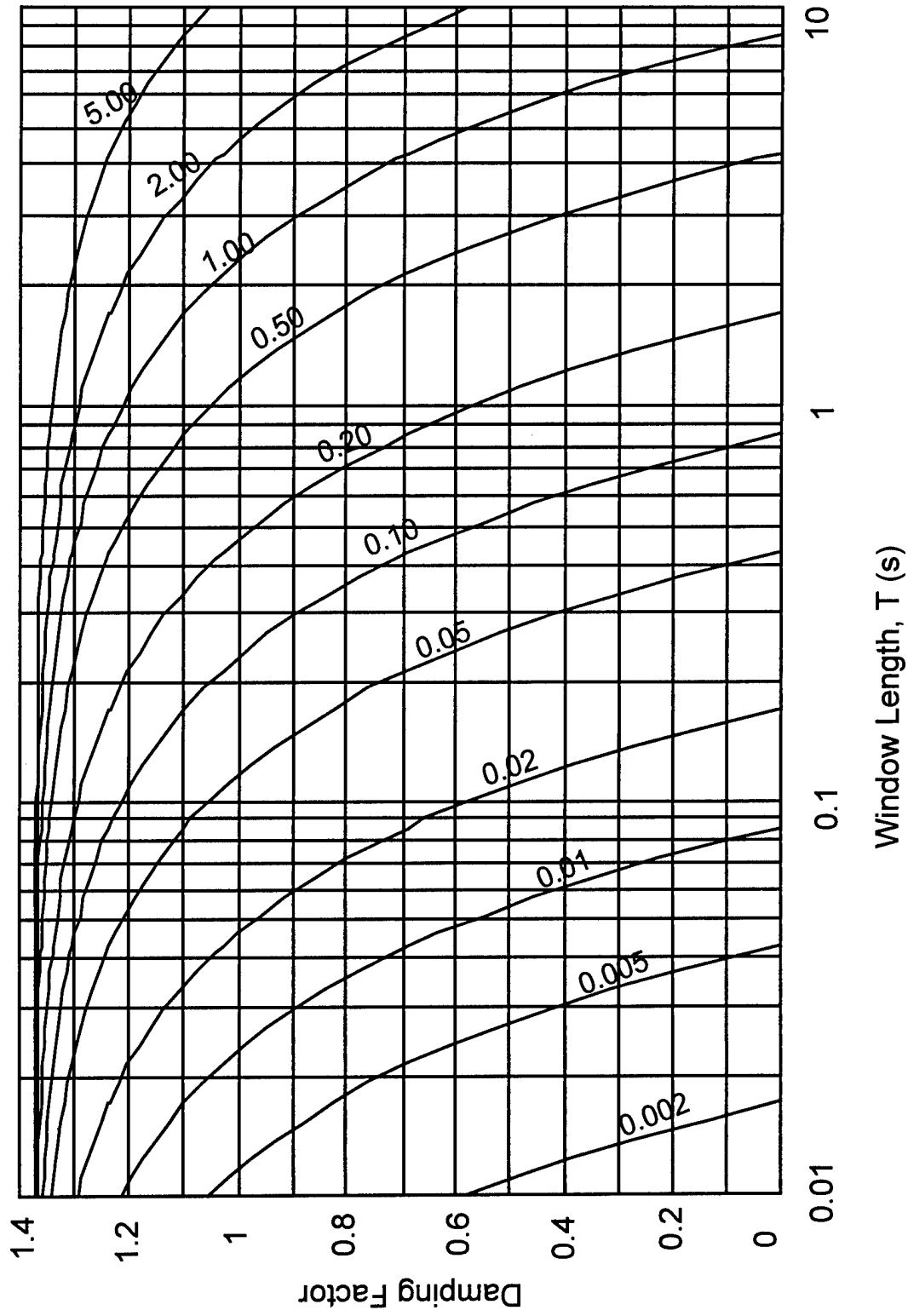
Figure 8. Nomograph to determine the exponential window time constant, based on the length of the windowed time record, and the type of structure. The number on the curved lines are the required exponential time constant in seconds. The nomograph is for an attenuation of –75dB. See Section 6.8.5.
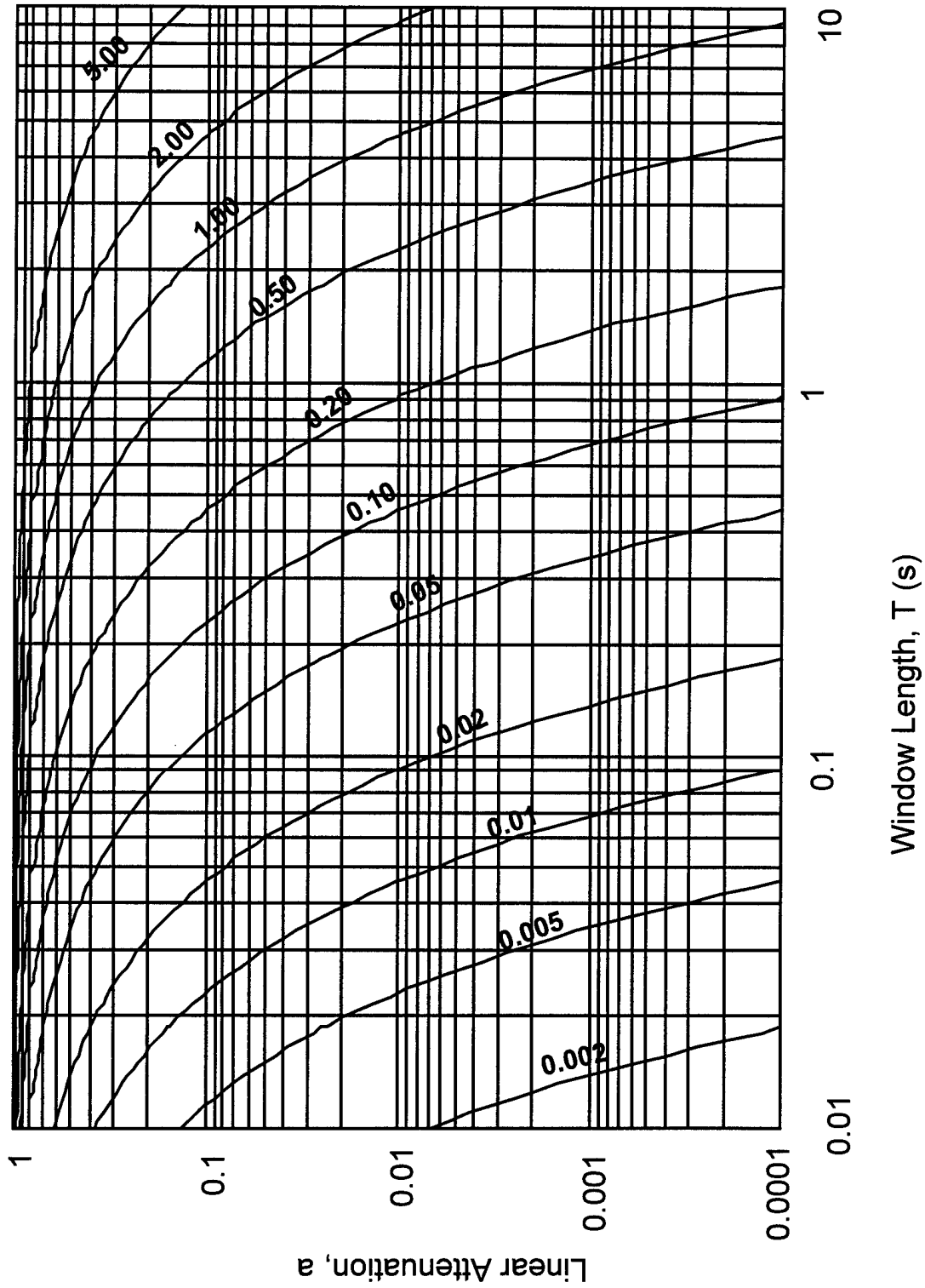
Figure 9. Nomograph to convert an exponential window time constant to an exponential window attenuation, based on the length of the windowed time record. The numbers on the curved lines are the exponential time constant in seconds.

7. Data Analysis

7.1 General

7.1.1 This standard does not cover the detailed data analysis and reduction necessary to obtain the modal damping estimates. However, a general summary is in order. Also, there is an important correction that has to be made to the results obtained from the modal analysis. This correction is included in Section 7.2 of this standard.

7.1.2 The general procedure for a modal analysis falls in two main areas. The first is to identify the natural frequencies and modal damping, and the second is to determine the mode shapes. Some commercial packages require two passes of the data to complete the analysis, while others can determine all the modal parameters in a single pass. When there is a choice, in general, a two-pass procedure is preferred.

7.2 Correction to Modal Damping

7.2.1 The windowing necessary to obtain good Fourier transforms also introduces extra damping into the data, which is not removed by the analyzer hardware. This extra damping is therefore included in the damping estimate made during the modal analysis. Hence, the damping values obtained from the modal analysis have to be corrected. The correction is a function of the window time constant, τ, and the structural natural frequency, f_n (Hz). For estimates of viscous damping ratio obtained from a modal analysis of the windowed data, ξ_M, the corrected viscous damping ratio, ξ_C is determined as:

$$\zeta_C = \zeta_M - \left(\frac{1}{2\pi f_n \tau}\right) \infty 100\%$$

8. Conclusions

8.1 The experimental determination of structural or material damping as a function of frequency can be a complicated task. Often the values determined from different test methods, or by different facilities, can be significantly different. The standardized procedure presented here aims to reduce these variations.

8.2 The standard focuses on the four main areas where variations in test procedure can introduce differences in observed damping estimates. The areas are: a) Preparation and support of the structure; b) selection, preparation and use of transducers and equipment; c) setting up the analyzer, and d) data analysis and correction.

9. References

Ewins, D.J., *Modal Testing: Theory and Practice*, Research Studies Press Ltd., Letchworth, Hertfordshire, England, 1986.

Halvorsen, W.G. and Brown, D.L., "Impulsive Technique for Structural Frequency Response Testing," JSV 8-21 Nov 1977

Ratcliffe, C.P., *Correcting a Significant and Consistent Error in the Modal Damping Obtained using Transient Vibration Data*, USNA-TR-98-02, April 1998.

Glossary

Accelerance – Accelerance is the dynamic function that defines the acceleration response of a structure per unit force. It is a complex function of frequency.

Apparent mass – Apparent mass is the dynamic function that is the reciprocal of accelerance.

Dynamic frequency response functions – The dynamic functions typically required for a modal analysis are accelerance, mobility or receptance. Other dynamic functions often measured during a vibration trial, but not usually used during modal analysis, are dynamic stiffness, mechanical impedance, apparent mass, and transmissibility.

Dynamic stiffness – Dynamic stiffness is the dynamic function that is the reciprocal of receptance.

Mechanical impedance – Mechanical impedance is the dynamic function that is the reciprocal of mobility.

Mobility – Mobility is the dynamic function that defines the velocity response of a structure per unit force. It is a complex function of frequency.

Receptance – Receptance is the dynamic function that defines the displacement response of a structure per unit force. It is a complex function of frequency.

Transmissibility – Transmissibility is the dynamic function that defines the response of one point of a structure relative to the response at another. For vibration mounts, transmissibility can also be the ratio of the force transmitted to a foundation relative to the dynamic force creating motion. Transmissibility is a complex function of frequency.